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PREDICTING OF STEAM CONDENSATION HEAT TRANSFER COEFFICIENT IN HORIZONTAL FLATTENED TUBE

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Abstract: The heat transfer coefficient of steam condensation has a significant role in the performance of air-cooled heat exchangers. The purpose of this work is to predict the local/average local steam condensation heat transfer coefficient inside the horizontal flattened tube under vacuum conditions using numerous correlations that were developed by some researches which have been conducted under specified conditions. The results from these correlations have been compared with experimental data of Davies, therefore more investigate for the values are necessary to improve or/and validate the existing correlations. The effect of such parameters like the uniform heat flux and saturation temperature also have been studied on the local steam condensation heat transfer coefficient as the results show that the heat transfer coefficient decrease as the heat flux increase, while it increases as the steam saturated temperature increase.

Keywords: *Condensation, Flow Regime, Heat transfer Coefficient*

1. Introduction

Many Industries applications include condensation phenomenon inside horizontal tubes such as air-cooled power generation plants, desalination, air-conditioning refrigerant and Concentrated Solar Power [1][2][3]. Substantial importance to understand the flow patterns and heat transfer process that will lead to the development of the purpose of the systems precisely and in-deepness. Even though circular cross-section tubes are utilized in the most of these applications, there are in some specific state requiring to employ non-circular tubes (elliptic cross section [4], rectangular cross section [5]and flattened cross section tubes [6]as in air-cooled steam condenser.

2. Methods and Mathematical Formulation

2.1. Flow Regimes

There is correspondence in the literature that the pressure drop and heat transfer mechanisms are closely related to the predominant two-phase flow system. For appropriate examine, various kinds of relationship are required to be done for guess in dissimilar condensation flow regimes, and the shortage to determine the proper kind can result in intense errors. This leads to the deduction that determination of the flow regime is one of the most important(relevant)

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proceedings in proper guess of condensation heat-transfer coefficient inside horizontal tubes.

the interaction impact between buoyancy, gravity, inertia and surface tension forces leads to a convert in visible shape of the flow patterns and change the magnitude of void fraction and pressure drop.

Flow regime guess is significant and complicated in phase change operations since the vapor fraction varies during the flow in the tube therefore the flow patterns may change between inlet and outlet of the tube.

Many flow regime maps have been submitted over the years which depend on the quality of vapor, flow rate, tube diameter, and fluid properties for guessing two-phase flow regime transitions in horizontal tubes. The maps of Taitel and Dukler [7]shows condensation route for R-134a with varies range from 5 to 95% quality at steps 5%, and Baker [8] for horizontal flows is widely utilized in the petrochemical industry are may be those most prominent.

The criterion of dominant flow inside the air-cooled steam condenser tubes are changeable as a result of changing operational situations and that is why it's obligatory to define the dominant criterion for subsequently move to the analyzing of the case. Martinelli-Lockhart[9] is used as a parameter to solve this problem which depends on the gathering of two dimensionless parameters as in (1),(2):

Martinelli's parameter

$$X_{tt} = \left(\frac{1-x}{x}\right)^{0.9} \left(\frac{\rho_v}{\rho_l}\right)^{0.5} \left(\frac{\mu_v}{\mu_l}\right)^{-0.1} \quad (1)$$

Dimensionless speed:

$$J_g = \frac{xG}{\left(gd\rho_v(\rho_l-\rho_v)\right)^{0.5}} \quad (2)$$

After the implementation of equations (1) and (2), it is feasible to determine the flow pattern dominant by using quantities that given in table (1):

Table (1). The ranges for the flow pattern dominant condensation parameters in air-cooled steam condenser.

Validation Parameters	Flow Model
$J_g > 1.5$; $X_{tt} < 1$	Annular
$J_g \leq 1.5$; $X_{tt} < 1$	Stratified-wavy
$J_g \leq 1.5$; $X_{tt} \geq 1$	Intermittent
$J_g > 1.5$; $X_{tt} \geq 1$	Bubbled

2.2 Void Fraction

Void fraction (e) or may could liquid holdup (1-e) is the volume of space occupied by the gas or liquid which used to predict the flow regime transitions, pressure loss and heat transfer that considered as unknown parameters in any two-phase system. There are many models or correlations to calculate or predict the void fraction that can be classified as follows [10]:

- Homogenous model (depend on assumption that liquid and gas phases move at same velocity).
- Slip ratio models (main frame was put by Butterworth (1975) based on the ratios of quality, densities and viscosities of liquid and gas phases.
- $K_{\varepsilon H}$ correlations which represented by a constant or function multiple of the homogenous void fraction.
- Drift flux model (In this model take into account the non-uniformity in the flow)
- General Correlations (most of these correlations are empirical in nature with basic implicit principles combined into the

various parameters when developing them).

Obviously, the problem is to define the phase distribution from input states in a specified pipe is more difficult as a result of the slippage between the two phases (gas and liquid).

consequently, it is necessary to use a mode is valid and dependable over the whole range of flow regimes, mass velocities, reduced pressures and mass velocities. therefore, simple logarithmic mean void fraction (LMε) between the rangrs of homogenous and non- homogenous void fraction was provided a best result, where the logarithmic mean void fraction Ɛ is defined as [7]:

$$\varepsilon = \frac{\varepsilon_h - \varepsilon_{ra}}{ln\left(\frac{\varepsilon_h}{\varepsilon_{ra}}\right)} \quad (3)$$

$$\varepsilon_h = \left[1 + \left(\frac{1-x}{x}\right)\frac{\rho_v}{\rho_l}\right]^{-1} \quad (4)$$

$$\varepsilon_{ra} = \frac{x}{\rho_v}\left([1 + 0.12(1-x)]\left[\frac{x}{\rho_v} + \frac{1-x}{\rho_l}\right] + \frac{1.18(1-x)[g\sigma(\rho_l - \rho_v)]^{0.25}}{G\rho_l^{0.5}}\right)^{-1} \quad (5)$$

The major task that is faced design engineer is to choose the convenient correlation between a large number of correlations available. the majority of the correlations have some makeup of constraints linked them. The thought of collecting several correlations that declare to work well for certain flow characteristics for which the researchers are interested in attempting to develop all the design requirements.

2.3 Heat Transfer Coefficient

Many models for local or local average condensation heat transfer coefficient have been proposed. That early were labeled in two types of flow pattern stratified flow where the gravity dominated flow has been molded as a thick layer of condensate flowing in the lower part of the tubes whereas a thin film of liquid cover the rest wall of the upper part of the tubes and annular flow where shear dominated, two various ideas were considered for determine the heat transfer coefficients: shear stress between liquid and gas phases or the correlations of two-phase multiplier which prevalent (commonly) used to calculate the Nusselt number of the condensation phenomenon (Nuesslt number of single phase for turbulent mode) by multiplying it with convenient two-phase multiplier.

Generally, the two-phase multiplier assumed as a function of density ratios between the phases (liquid and vapor), viscosity, vapor quality and, reduced pressure, liquid Froude number, Martinelli (X_{tt}) parameter, etc.

Akers et al. [11][12] submitted a set of equations in the ASHRAE Handbook is a design tool that covers the annular and stratified flow conditions and usable at certain ranges of convenient dimensionless parameters. Kosky and Staub [13] and Traviss et al. [14], developed their own procedures of calculation by assumed that the Von Karman velocity profile for pipe flow fills in the condensate annulus truly and valid only with an annular flow pattern. Chato [15] submitted a set of equations for stratified and annular flow regime that able to predict the coefficient of heat transfer; the correlation for annular flow was derived using the two-phase multiplier idea. Cavallini et al. [16] examined some methods to predict the condensation heat transfer coefficients of refrigerants (halogenated type) inside horizontal tubes, for experimental data set got by independent researchers. The results of this exam showed the ability to predict suitably the experimental data, that is mostly within +-20%.

Shah [17] is presented a simple dimensionless correlation for predicting HTC during film condensation inside pipes. It has been tested by compared to different experimental data that condensed in pipes for ranging diameter with various orientation. The mean deviation for these data was found to be 15.4%. This correlation was improved to cover low mass flux and widen the range of applicability [18]. The result showed good agreement for mass flux (4 to 820 kg/m^2. s).

Shen et al [19] measured the circumferential distributions of local condensation heat transfer coefficient and wetted angle through, in stratified flow and vacuum conditions of steam condensation in a horizontal tube. The accurate of experimental wetted angle is predicted by using the Biberg correlation and Rohani void fraction correlation. The condensation on wall of tube is divided into film wise section (upper part) and accumulate section (bottom part). The local HTC in film wise condensation section is obviously higher than liquid accumulation section. A new Correlations for local steam condensation heat transfer coefficient in stratified flow are proposed depend on experimental data.

For pure vapors at saturated conditions, many correlations of local condensation heat transfer coefficient have been proposed under forced convection conditions inside horizontal tubes. The following correlations were used to predict the HTC depend on

2.3.1 *Shah Correlation for Horizontal Tubes*

Shah correlation [15] is valid according to the following range of conditions:

- Mass flux range from 4 to 820 kg/m^2. s
- Tube diameter from 2 to 49 mm
- Fluids are steam, refrigerants, methanol, ethanol, propane, propylene and isobutene.
- Geometry and orientation: circular, vertical, inclined and horizontal
- Prandtl number range: 1to 18

According to data analysis, the criteria bounded between heat transfer zones:

$$J_g \geq 0.98(Z+0.263)^{-0.62} \quad (6)$$

Zone 1 exist if

$$J_g \leq 0.95(1.254+2.27Z^{1.249})^{-1} \quad (7)$$

Zone 3 exist if

While zone 2 existed between zones 1 and 2.

J_g is the dimensionless velocity of vapor as mentioned in equation (2), Subsequently Shah [20] given flow pattern based on the above correlation:

For intermittent, annular and mist flow can use

$$h_{TP}=h_{Nu} \quad (8)$$

For wavy flow can use

$$h_{TP}=h_l+h_{Nu} \quad (9)$$

For stratified flow can use

$$h_{TP}=h_l \quad (10)$$

where h_l and h_{Nu} as below

$$h_l = h_{ls}\left(1+\frac{3.8}{Z^{0.95}}\right)\left(\frac{\mu_l}{14\mu_g}\right)^{(0.0058+0.557p_r)} \quad (11)$$

$$h_{ls} = 0.023Re_{ls}^{0.8}Pr_l^{0.4}k_l/d \quad (12)$$

$$Re_{ls} = G(1-x)d/\mu_l \quad (13)$$

$$h_{Nu} = 1.32Re_{ls}^{-1/3}\left[\frac{\rho_l(\rho_l-\rho_g)gk_l^3}{\mu_l^2}\right]^{1/3} \quad (14)$$

Worth mentioning, Shah [15] showed that zone 3 is expected at very low flow rates and the data were not available to analyze.

2.3.2 Thome & Cavallini Correlation

The idea of the new model starts from the similarity between annular film evaporation and annular film condensation inside tubes [21] by assuming a uniform thickness of liquid around the whole internal perimeter of the tube.

The flow structure for evaporation that assumed by Kattan et al [22] can be utilized to condensation which the upper part of the tube in a stratified flow will be wetted by film of liquid instead of remain dry during evaporation. Therefore, the new model suggests by Thome et al. suppose three geometries characterizing fully stratified wavy flow, stratified-wavy flow and annular flow.

The objective of Thome et al. is to develop a new model for heat transfer to predict local HTC involve numerous empirical parameters.

Two kind of heat transfer mechanism take place within the tube (as below in figure (1)) in condensation model: convective condensation and film condensation.

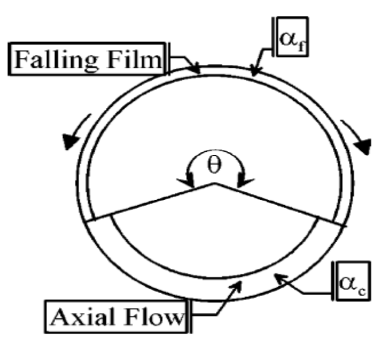

Figure. 1. Falling film and Convective in heat transfer model

The heat transfer coefficient of convective condensation hc is used to the perimeter wetted by the liquid film which flows axially, which refers only part of the perimeter in stratified-wavy and fully stratified flows (bottom part), while to the entire perimeter in annular, intermittent and mist flows. The liquid film flow is supposed to be turbulent along the axis. The film condensation heat transfer coefficient hf is applied to upper perimeter of the tube for stratified-wavy and fully stratified flows. The Nusselt falling film theory is applied to obtain hf inside of the horizontal tube, which proposes the falling film is laminar.

The global term of local heat transfer coefficient h_{TP} at any cross section of the tube is:

$$h_{TP} = \frac{h_f r\theta + (2\pi - \theta) r h_c}{2\pi r} \qquad (15)$$

Where r is the radius of the internal and θ is the falling film angle around the upper part of perimeter of the tube which can determine as flows:

$$\theta = \theta_{strat}\left\{\frac{(G_{wavy} - G)}{(G_{wavy} - G_{strat})}\right\}^{0.5}$$

(16)

And θ_{strat} can be expressed according to Biberg [23]

G_{wavy} and G_{strat} are specified in accordance with

$$\theta_{strat} = 2\pi - 2\left\{\begin{array}{l}\pi(1-\varepsilon) + \left(\frac{3\pi}{2}\right)^{1/3}\left[1 - 2(1-\varepsilon) + (1-\varepsilon)^{1/3} - \varepsilon^{1/3}\right] \\ -\frac{1}{200}(1-\varepsilon)\varepsilon\left[1 - 2(1-\varepsilon)\right]\left[1 + 4\left((1-\varepsilon)^2 + \varepsilon^2\right)\right]\end{array}\right\} \qquad (17)$$

[10]

$$G_{wavy} = \left\{\frac{16A_{vd}^3 g d \rho_l \rho_v}{x^2\pi^2(1-(2h_{ld}-1)^2)^{0.5}}\left[\frac{\pi^2}{25h_{ld}^2} * \left(\frac{We}{Fr}\right)^{-1.023} + 1\right]\right\}^{0.5} + 50 - 75e^{-\left(x^2-0.97\right)^2/x(1-x)}$$

(18) (19)

$$G_{strat} = \left\{\frac{(226.3)^2 A_{ld} A_{vd} \rho_v(\rho_v - \rho_l)\mu_l g}{x^2(1-x)\pi^3}\right\}^{1/3} + 20x$$

The heat transfer coefficient of film condensation h_f is got from the theory of Nusselt for laminar flow of upper part of the internal perimeter of the tube (falling film part), where h_f is the mean coefficient for this perimeter. Instead of integrating from the top of the tube to the stratified liquid layer at θ/2 to obtain h_f ,which would be more satisfactory in theoretical work.

Then h_f

$$h_f = 0.728 \left[\frac{g\, h_{fg}\, k_l^3\, \rho_l\,(\rho_l - \rho_v)}{\mu_l\, d\,(T_{sat} - T_w)}\right]^{1/4} \quad (20)$$

Typically, a heat flux is assumed in the deigning of heat exchanger that implemented in every incremental zone therefore it is suitable to change the correlation in equation (19) to heat flux utilizing Newton's low of cooling. Then h_f is given below

$$h_f = 0.655 \left[\frac{g\, h_{fg}\, k_l^3\, \rho_l\,(\rho_l - \rho_v)}{\mu_l\, d\, q}\right]^{1/3} \quad (21)$$

For the lower part of tube which convective condensation heat transfer coefficient hc(turbulent film equation), the expression is followed below

$$h_c = \frac{c\, Re_{l\delta}^{n} Pr_l^{m} k_l f_i}{\delta} \quad (22)$$

Where $Re_{l\delta}$ is the liquid film Reynolds number depend on the mean liquid velocity of the liquid in liquid cross-section area as

$$Re_{l\delta} = \frac{4G(1-x)\delta}{(1-\varepsilon)\mu_l} \quad (23)$$

The best values for exponents: (c=0.003), (n=0.74), (m=0.5) for turbulent falling film condensation on a vertical plate.

δ is the liquid film thickness for annular flow thus can be calculated as

$$\delta = \frac{d(1-\varepsilon)}{4} \quad (24)$$

The interfacial surface roughness was verified as being a specific convective condensation effects for the following reasons:

1- The liquid film across the interface is transmitted due to the shear of high speed of the vapor subsequently increases the magnitude and number of the waves created at the interface, which in turn increases the available surface area for condensation, tending to increase heat transfer.
2- The waves at the interface are not continuous(non-sinusoidal) and consequently the mean thickness of the film is tending to reduce, again increasing heat transfer.

For these two reasons, the interfacial roughness correction factor f_i was introduced to act on h_c for fully stratified flow

$$f_i = 1 + \left(\frac{u_v}{u_l}\right)^{1/2} \left(\frac{g\delta^2(\rho_l - \rho_v)}{\sigma}\right)^{1/4} \left(\frac{G}{G_{strat}}\right) \quad (25)$$

Also has been paralleled to test data of fifteen refrigerant fluids and examined over the range of conditions as below:

- Range of Mass flux 24 to 1022 kg/m^2. s
- Vapor quality are from 0.03 to 0.97
- Maximize reduced pressure 0.02 to 0.8
- Range of internal tubes diameters are from 3.1 to 21.4 mm.

2.3.3 *Shen Empirical correlation*

Shen et al [19] estimated the circumferential distributions of local heat transfer coefficient and wetted angle as shows in Fig.2 across the stratified flow of steam condensation in horizontal tube under vacuum conditions.

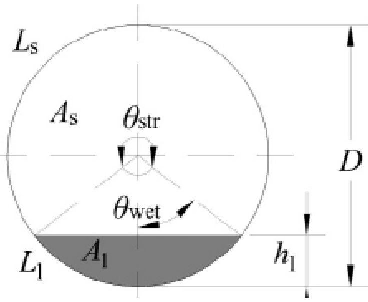

Figure 2. Geometrical parameters in stratified flow

The previous correlations of Chato's [24],Jaster [25] and Singh [26] are related to average heat transfer coefficient based on the experimental data of refrigerants condensation heat transfer and the minimum mass flux is 20(kg/m^2.s), as steam mass flow rate is no more than 10 (kg/m2.s) hence using these correlations to satisfy the estimation of steam condensation heat transfer coefficient are difficult. Shen correlations can calculate the local heat transfer coefficient at any location on the tube wall in the liquid accumulation section depend on data that obtained from his experimental work and utilizing the forms of Dobson's correlation. These correlations be invented the effect of steam quality, physical properties and mass flow rate in the liquid accumulation section and temperature difference in filmwise condensation section.

The local steam condensation heat transfer coefficient of stratified flow at any angle along the circumferential direction is determined by (26), (27):

In the filmwise section

$$Nu_{up} = \frac{0.02Re_{so}^{0.318}}{1+1.11X_{tt}^{0.755}}\left(\frac{gh_{fg}D^3\rho_l(\rho_l-\rho_v)}{k_l\mu_l(T_s-T_w)}\right)^{0.25} \qquad (26)$$

$$Re_{so} = \frac{Gd}{\mu_g} \qquad (27)$$

In the section of liquid accumulation

$$Nu_{bot,0} = 0.033Re_{ls}^{0.8}Pr_l^{0.33}\left[1 + \left(\frac{\rho_l}{\rho_v}\right)^{0.5}\left(\frac{x}{1-x}\right)\right]^{0.8}\left(\frac{\cos\theta_{wet}}{\cos\theta}\right)^{1.31} \qquad (28)$$

3. Discussion

3.1 Heat Transfer correlations comparison

Figure (3) shown the comparison of results predicting steam condensation heat transfer coefficient for numerous correlations. The result of heat transfer coefficient is adopted by Thome et al. correlations are higher than the other values of correlations, The Thome et al. correlation can be implemented assuming uniform heat flux or constant temperature difference (wall temperature) along the condensing tube. in order to satisfy the requirements of actual operation conditions of air cooled steam condensers, the assumption of constant surface temperature invalid therefore uniform heat flux is convenient in terms to the default solution for the system.

According to Thome et al. correlations, the stratified flow model (or wavy-stratified flow), the vapor is characterized by low (or medium) flow rates, therefore the tube cross-section is divided into two sections, the upper section which occupied by vapor (has a high heat transfer coefficient) and the lower section where the liquid is accumulated (has low heat transfer coefficient) while the relationship between the two sections is the stratified angle. when vapor flow rates are increased the transition from stratified (or wavy-stratified) flow to annular flow is happened which is not desirable in heat exchangers using air as a coolant (like air-cooled condensers in power plants) because of liquid layer covered the wall acts a barrier between the vapor and the internal surface of the tube resulting to reduces the heat that transporting to air.

The results to predict the heat transfer coefficient with Shah correlation are lower than the values obtain from the Thome et al. correlations. Shah's relations are based on the hypothesis that the flow inside the tube is annular flow (not stratified flow). This can be seen by the absence of a stratified angle around upper part of the tube in the correlation. In his research (16), he pointed out the analyzable data for like these conditions were not available to expect the third zone where It has very low flow rates, which is suitable for air-cooled condensers used in power plants.

The consequences of local heat transfer coefficient using the Shen collations are less valuable than the rest of the results of the correlations, which divided the condensation process into upper part of wall

tube that film wise condensation and lower part of the tube that liquid accumulation.

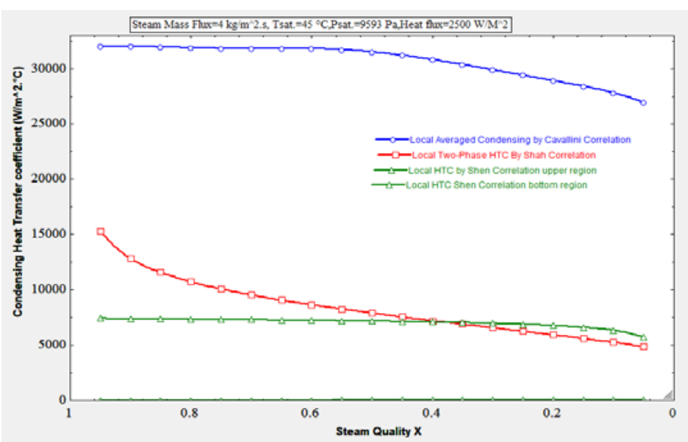

Figure 3. Results of predicting steam condensation heat transfer coefficient for numerous correlations

Figure (4) shows the comparison of results predicting steam condensation heat transfer coefficient for numerous correlations with Davies experimental data [27] for horizontal, (or inclined) flattened-tube steam condensation heat transfer coefficient more precise results for the values of heat transfer coefficient are necessary to improve or/and validate the existing correlations Thome's et al. correlations will be adopted to show the effect of the other parameters on steam condensation heat transfer coefficient.

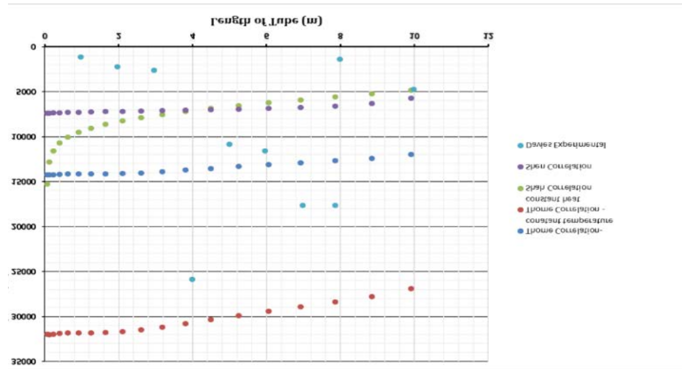

Figure 4. The comparison of results predicting steam condensation heat transfer coefficient for numerous correlations with Davies experimental data

3.2 Impact of uniform Heat Flux

Figure (5) shows the impact of the uniform of heat flux on the steam condensation heat transfer coefficient along with the variation of steam quality for G=4 kg/m2. s, $T_{sat.}$ =45°C and flattened tube dimension (H=0.2113 m, W=0.0163 m). The plot shows that the condensation heat transfer coefficient decrease with increases in the uniform heat flux. At the beginning Increasing the uniform heat flux, leads to maximize the rate of heat transfer that extracted from the steam and increases the speed of formation of liquid film on the tube wall. This liquid film acts as a barrier between the hot vapor and the cold tube wall, which results in a lower condensation heat transfer coefficient.

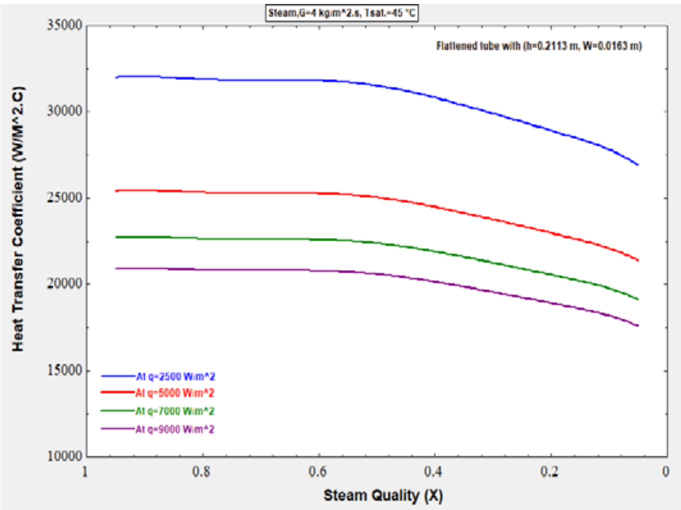

Figure 5. The effect of the uniform of heat flux on the steam condensation heat transfer coefficient along with the variation of steam quality

3.3 Impact of saturation temperature

Figure (6) shows the impact of saturation temperature on the steam condensation heat transfer coefficient along with the variation of steam quality for G=4 kg/m2. s, q =2500 W/m^2 and flattened tube dimension (H=0.2113 m, W=0.0163 m). The plot shows that the condensation heat transfer coefficient increase with increases in the saturation temperature of steam. When the saturation temperature increases, the thermal conductivity of the vapor and liquid phases will get better and the viscosity of the water condensate (film liquid) decreases. This will accelerate the slippage of the liquid film formed by the condensation process around the upper part of the pipe to the lower part leads to reduce the thickness of the liquid film is at the top, so the heat transfer coefficient increases with the steam saturation temperature.

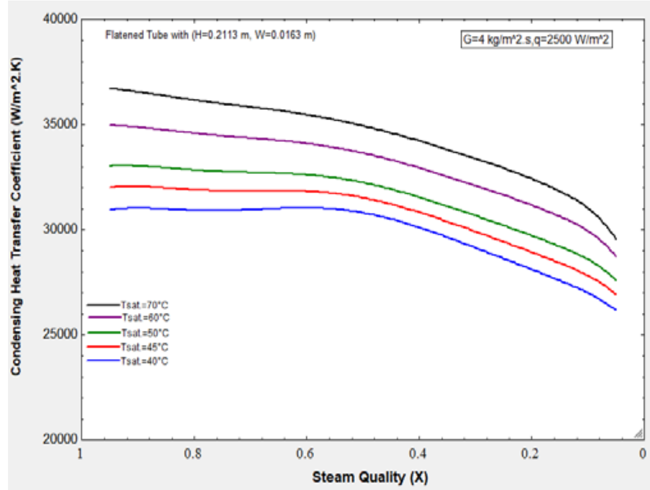

Figure 6. The effect of saturation temperature on the steam condensation heat transfer coefficient along with the variation of steam quality

4. Conclusions

In this work, the comparison between numerous of predicting local or average local heat transfer coefficient of steam condensation inside along fattened horizontal tube and the effect of such parameters like the uniform heat flux and saturation temperature on the steam condensation heat transfer coefficient. The following items can be concluded:

1- To predict condensation heat transfer, more precise results for the values are necessary to improve or/and validate the existing correlations.
2- The condensation heat transfer coefficient decrease with increases in the uniform heat flux.
3- The condensation heat transfer coefficient increase with increases in the saturation temperature of the steam.

5. Acknowledgements

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6. Nomenclature

A tube cross-section area , m^2
A_l liquid cross-section area ,Al=A(1- ε), m^2
A_{ld} dimensionless cross-sectional area occupied by liquid, $A_{ld}=A_l/d^2$
A_V vapor cross-sectional area ,Av=A ε ,m^2
A_{vd} dimensionless cross-sectional area occupied by vapor , $A_{vd}=A_v/d^2$
c convective film constant [–]
c_{pl} specific heat of liquid [J/(kg.K]
d inner tube diameter [m]
f_i factor of interfacial roughness [–]
g acceleration of gravity (m/s2)
G total mass flow rate (mass velocity) of liquid and vapor [kg/(m^2.s)]
G_{mist} transition mass velocity of mist flow [kg/(m^2.s)]
G_{strat} transition mass velocity of stratified flow (kg/(m^2.s))
G_{wavy} transition mass velocity of wavy flow (kg/(m_2.s))
h_{fg} condensation latent heat, [J/kg]
h_{ls} heat transfer coefficient assuming liquid phase flowing alone in the tube
h_l heat transfer coefficient assuming all mass flowing as liquid
h_{TP} two-phase heat transfer coefficient
h_{Nu} heat transfer coefficient, the Nusselt relation
h_f Nusselt film condensing coefficient on top perimeter of tube (W/(m^2.K)
h_c convective condensation coefficient, (W/m^2.K)
j exponent (j ¼ 1=2) [–]
k thermal conductivity, [W/(m.K)]
k exponent (k ¼ 1=4) [–]
m exponent on Pr [–]
n exponent on Re [–]
P pressure, [N/m^2]
Pr Prandtl number
Q heat flux from fluid to tube [W/m^2]
r inside radius of tube (m)
Re_l Reynolds number of liquid film [–]
T temperature [K]
u_l mean liquid velocity in film, u_l= G(1- ε)/(ρ_l(1- ε)), [m/s]
u_v mean vapor velocity, u_v= Gx/(ρ_v ε),[m/s]
x vapor quality [–]
X_{tt} Martinelli parameter with both phases turbulent [–]

Greek symbols

δ thickness of liquid film as annular ring (m)
ε vapor void fraction [–]
ε_{ra} Rouhani-Axelsson void fraction [-]
μ_l dynamic viscosity [N s/m^2]
θ angle of the upper portion tube not wetted by stratified liquid [rad]
θ_{strar} stratified angle about upper perimeter of the tube [rad]
θ_{wet} wetted angle
ρ density [kg/m3]
σ surface tension [N/m]

Subscripts

l liquid
g vapor
w wall
sat saturated
r reduced
crit critica

Conflict of interest

There are not conflicts to declare.

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